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Multiply Connected Cavity Flows in Turbomachines

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ABSTRACT

Changing any one component in a set of multiply connected cavities, such as the compressor discharge seal, causes flow changes throughout the entire engine. T–700 engine test data and numerical simulations of the space shuttle main engine high-pressure fuel turbopump, the four-stage T–56/501D turbine, and swirl brake dynamics are cited as examples. The needs for time-dependent optimization of turbine and compressor flow fields, for use of seals to enhance rotor stability, for predictive maintenance, and for a clean-sheet approach to engine design are concluded.

Keywords: Seals; Cavity flow; Turbine; Turbomachine compressor

1. INTRODUCTION

Fluid flows within an engine or turbomachine are coupled. The power stream and the secondary flow streams that cool components and function as working or purge fluids are coupled through the seal and cavity flow fields (Hendricks et al., 1995). The behavior of these flows becomes critical to the thermal management of the component and ultimately to the performance of the turbomachine.

As turbomachine power systems mature, the ability to refine component efficiencies declines and efforts focus on small percentage gains. In conventional turbomachine analyses interactions between the power stream and the secondary flow paths, such as those beneath the blade platforms, beyond the blade and vane tips, around the diffuser, and in the combustor sections, are not coupled. For large changes in efficiency this approach is valid; however, for small changes the coupling becomes of major significance. Current numerical and experimental work is focused on determining the interaction between the power stream and multiply connected multicavity sealed secondary flow fields and on the conjugate thermomechanical response.

2. COMPRESSOR DISCHARGE

Under a U.S. Army/NASA YT–700 engine test program it was realized that large gains in engine efficiency could be realized for a small investment in sealing technology (Hendricks et al., 1995). In a "piggyback" test the five-forward-facing-labyrinth-tooth compressor discharge (CDP) seal was changed out to a dual-brush CDP seal (standard Cross construction with chromium carbide rub runner) that nominally decreased leakage by a factor of 2. Although the original leakage flows were only of the order of 0.1% of the power stream flow, the improvement in engine specific fuel consumption was over 1% (Fig. 1). The reason is that the brush seal increased the compressor pressure ratio, which changed flows thoughout the engine. Changing the internal flow field only a little changed the power stream flow in the compressor, combustor, and turbine (i.e., the rest of the engine), and this is the most important result of the test.

3. MULTIPLE-CAVITY FLOW

The YT–700 seal test results led to a detailed investigation of the flow fields in the Daniels and Johnson (1993) United Technologies Research Center (UTRC) simulation of the space shuttle main engine (SSME) high-pressure fuel turbopump (HPFTP) (Fig. 2). It was clear that first-order results could be obtained when using the boundary conditions specified by the Daniels and Johnson UTRC data for individual purge and cavity flows but that the concentrations were only qualitatively matched. Closer examination of the flow details revealed severe discrepancies between single and connected cavity flows. In fact, a satisfactory solution could not be obtained until all the cavity details were gridded and connected, all the purge flow boundary conditions were set, and the power stream boundary conditions were set at the pump inlet and exit (Athavale et al., 1995a). Comparing these numerical results with results for carbon dioxide tracer gas suggested the following:

1. Multiple-cavity analyses capture interactive power and secondary flow stream effects that cannot be realized for uncoupled single-cavity treatments; in short, uncoupled results are inadequate for determining small changes in performance. For the UTRC simulation of the SSME HPFTP a flow thread (Fig. 3) was defined that works its way through the upstream first-stage rotor seal and throughout the dual rotor cavities and exits downstream of the second-stage rotor seal.
2. Generally, there was good agreement with the experimental results for gas ingestion and flow egress, although the egress was lower than calculated.
3. Comparisons of predictions with the carbon dioxide concentrations in the central cavity regions was good, but agreement was only fair at the blade shanks. The shanks were simulated as rotating slots, although such simulations are known to be inadequate as the three-dimensional flow field is complex with spiral and angled jetting from the holes. A second reason for disagreement could be poor scanning of the prints for gridding.
4. The purge cooling thread (Fig. 3) tended to provide cooling for the front side of rotor 1 and aft the side of rotor 2. The ingested gas affected the aft part of rotor 1 and the front side of rotor 2. Such nonuniform cooling and heating leads to nonuniform stress in rotors.

4. FOUR-STAGE TURBINE

Ho et al. (1996) numerically studied flows in the entire inner portion of the T–56/501D engine turbine, including interstage cavities and seals interacting with the power stream. Their energy equation also includes the solid components. Solutions relating to thermal management usually face difficulty in specifying wall thermal boundary conditions. By solving the entire turbine section, this difficulty is removed and realistic thermal conditions are provided on the internal walls. The conjugate heat transfer analysis illustrates thermal gradients in the stationary intercavity walls as well as in the rotating disks that could lead to thermal strain distortions within the disks and within seal clearances. The simulations were performed using SCISEAL, an advanced three-dimensional computational fluid dynamics code for predicting fluid flows and the forces in turbomachinery seals and secondary flow elements.

The Allison T–56/501D turbine section model (Fig. 4) specifies the main flows; shaded areas represent solid material (conjugate heat transfer). Flow purge fluids are specified, and the structures labeled "stator support" represent the vane, cavity, or seal and in some cases also represent balance piston arrangements. The first rotor is the left boundary and the fourth rotor is the right boundary of Fig. 4. The rotor speed was held constant at 14 239 rpm; positions 2, 3, and 5 represent cooling flows. The turbine section modeled was blocked to facilitate computation as shown in Fig. 5. Under steady conditions the thermal balance was good. Figure 5 also illustrates the local static temperature details for stage 1–2 cavities (stator between rotors 1 and 2). The power stream ingestion into the cavity, although significant, does not cause irregular thermal contours within the cavity; however, the conjugate temperature gradients at the surface and within the stator disk are significant.

Figure 6 illustrates temperature details within the stage 2–3 and 3–4 cavities. Large radial and axial temperature variations are evident along the vane support for the interstage labyrinth seal between rotors 2 and 3 and to a lesser extent for the support between rotors 3 and 4, as was expected from the Allison design data.

5. ROTOR RUNOUT

Turbomachine rotation is eccentric to some degree, with whirl representing an extreme orbital motion that can destroy a machine. In a preliminary study of the T–56/501D 1–2 turbine interstage seal without conjugate heat transfer, Athavale et al. (1995b) fixed the clearances of the six-tooth-on-rotor labyrinth seal with a smooth stator at 0.012 and 0.024 in. The analysis showed that when labyrinth seal clearances are small (0.012 in.) turbine gas ingestion into the cavity is limited and cavity temperatures appear uniform, especially in the seal area (Fig. 7(a)). However, when the seal clearances are opened by operations or design to 0.024 in., the power stream flow is ingested into the cavity and heats the seal (Fig. 7(b)). If the rotor is whirling, the thermal distortions on the disk and the labyrinth seal teeth can lead to significant durability problems.

Athavale et al. (1992) found that when the rotor was offset one-half of the rim seal clearance into the power stream fluid ingestion was significant and that because the rotor was one-half of a rim seal clearance below the power stream fluid egress was large. Although coupled effects were not studied, the inference is that unsteady thermomechanical distress can become acute under dynamic loadings (see also Athavale and Hendricks, 1994).

Not addressed owing to the lack of data, but of major significance, are the acoustic fields generated and transmitted by the multiple cavities and their associated seals. This aeroacoustic behavior of fluid-generated instabilities in rotors has been noted in seals and should be part of the long-range seals code development program.

6. TIP AND RIM SEAL FLOWS

Tip and rim seals appear in both the compressor and the turbine, and they afford the highest payoff and biggest challenge and provide the greatest opportunity for a new engine. Current engine configurations require higher surface speeds, temperatures, and pressures with fewer turbine and compressor stages. These conditions tend to prohibit the use of unshrouded blades in high-pressure turbines. The work losses associated with tip seal losses can be 5%. Directing cooling air at the casing to reduce tip clearances works well, but it uses compressor air, adds a weight penalty, requires some maintenance, and has slow response to transients. Seal dynamics are usually not integrated into the engine analyses; however in our approach, seals become a major source of engine stability.

In a lid-driven flow blade simulation the blade flow carryover (tip seal losses) initiated a vortex on the opposite wall (pressure side) (Athavale et al., 1993). In visualization experiments using data and loss estimates provided by Moore and Tilton (1987) and Sholander and Amrud (1986), Bindon (1988) characterized the vortex as rolling from a conventional blade tip. Although these vortices are usually confined to the upper 10 to 20% of the blade, they represent a considerable performance loss through circumferential flow that performs no useful work and through disruption of the power stream (Denton, 1993). Bindon (1988) estimated losses based on linear turbine cascade results as 13% from endwall and secondary flow, 48% from mixing, and 39% from internal gap shear. Bindon also suggested streamlining tip geometries that represented a tradeoff between entropy production and flow turning. Note that the circumferential boundary conditions of the rotating shroud and the time-dependent vortex shedding structures (caused by blade passing) are not part of a linear cascade. Such vortices are influenced by rotor-stator interactions, and the strength and passage size are time dependent (Fig. 8; Yamamoto et al., 1993). The design of a radially bowed vane with modified airfoil contours showed a 50% reduction in total pressure loss relative to a radially straight vane (Fig. 9; Huber et al., 1985). Unsteady flow interaction between the tip seal vortex, blade and vane frequencies, and cavity ingestion of flow at the rim seal, although currently treated separately, are coupled flows that can excite the power stream. Counter- and co-rotating vortices could be introduced, to control vortex losses, noise sources, and stall margins, by controlled jets or swirl vanes in the stator platforms (Athavale et al., 1993).

7. SWIRL BRAKE ROTOR STABILITY

Destabilizing cross forces in unshrouded turbines originate mainly from the Alford mechanism (nonuniform work extraction due to varying blade efficiency with the tip gap). These forces increase substantially when the gap is reduced from 3% to 1.9% of the blade height. In shrouded turbines most forces arise from nonuniform seal pressures (Martinez-Sanchez et al., 1993).

Benckert and Wachter (1980) confirmed that the lateral force component (cross force) resulted from unsymmetrical pressure distributions in the eccentric gaps of labyrinth seals. To reduce these forces, the circumferential flows have to be reduced. Both effects, preswirl and shaft rotation, induce circumferential components. A suitable change in entry swirl ahead of the first labyrinth knife to oppose the shaft rotation can achieve a seal free of lateral (cross) forces. For short labyrinth seals the upstream "brake" is sufficient.

Childs and Ramsey (1990) experimentally determined the rotordynamic coefficient for a model of the alternative turbopump design of the SSME HPFTP interstage turbine seal. A 12-tooth labyrinth seal was run against a honeycomb stator. The swirl brake proved remarkably effective in reducing cross-coupled stiffness with increasing inlet or shaft velocity ratio. Childs and Ramsey concluded that the "proposed swirl brake design is remarkably effective."

Weiser and Nordmann (1990) cite the necessity for carefully assessing the labyrinth seal forces to determine the stiffness and damping of a turbomachine. They also note that the seals are often omitted from analyses owing to lack of computer programs to properly calculate their rotordynamic coefficients. They compared four models with data, with mixed results depending on the data set. Including swirl brakes did not substantially alter the first eigenfrequency but had a marked influence on the first forward bending mode.

These studies confirm the usefulness of swirl brakes in stabilizing labyrinth seal configurations and turbomachines in general. Engine rotordynamics are enhanced through good seal design.

8. TURBOMACHINE MAINTENANCE SYSTEM

Choy et al. (1996) have developed a method leading to engine maintenance scheduling and failure prediction. The technique includes identification of damage and wear, quantification of the damage level, and prognostication. The operating system can be modeled with a high degree of confidence; system diagnostic sensors and time-frequency vibration signatures are analyzed by using the Wigner-Ville distribution. Combining the model with the data base and predictions permits the operator to evaluate the engine and know if the system is degrading and whether the power loss is caused by gear malfunction or something else. With these results immediate steps can be taken to thwart failure.

9. CONCLUSIONS

From these studies of multiply connected cavity flows in turbomachines, we drew the following conclusions:

1. **Need to optimize coupling of multiply connected flow fields.** T–700 engine test results demonstrated that changing the compressor discharge seal from labyrinth to dual brush changed the flow fields throughout the engine and decreased specific fuel consumption by over 1%. Turbomachine test data and numerical predictions compared well only when the entire secondary flow field interacted with the power stream with boundary conditions set at the purge flows and upstream and downstream of the turbomachine. Thermal profiling of a four-stage T–56/501D turbine was validated only when the entire secondary flow field, with proper seal assessment, interacted with the power stream. Time-dependent flows and optimization require study.

2. **Need to use seals to enhance rotor stability.** Turbomachines and aeroderivative engines with labyrinth seals are often stabilized by inserting a swirl brake or redesigning with new seal rotordynamic data. In many aeroengines the interstage seals are labyrinths, and swirl brakes should be used to diminish the cross forces that contribute to rotor instability.

3. **Need to integrate maintenance predictions into analysis.** The combination of a historical data base, a model of the machine, and a wave ("sound") signature is proposed to provide predictive maintenance schedules. The value in these methods comes from knowing where, when, and why a failure will occur.

4. **Need a clean-sheet approach.** The entire secondary flow path and seal design should be reevaluated by using these numerical tools, experimental data, and design expertise. Those cooling and purge gases that are of marginal importance can be eliminated along with the structural distress associated with such holes. Further, this clean-sheet approach should be applied to all the associated components with thought given to active controls for a more efficient, long-life, low-maintenance engine.

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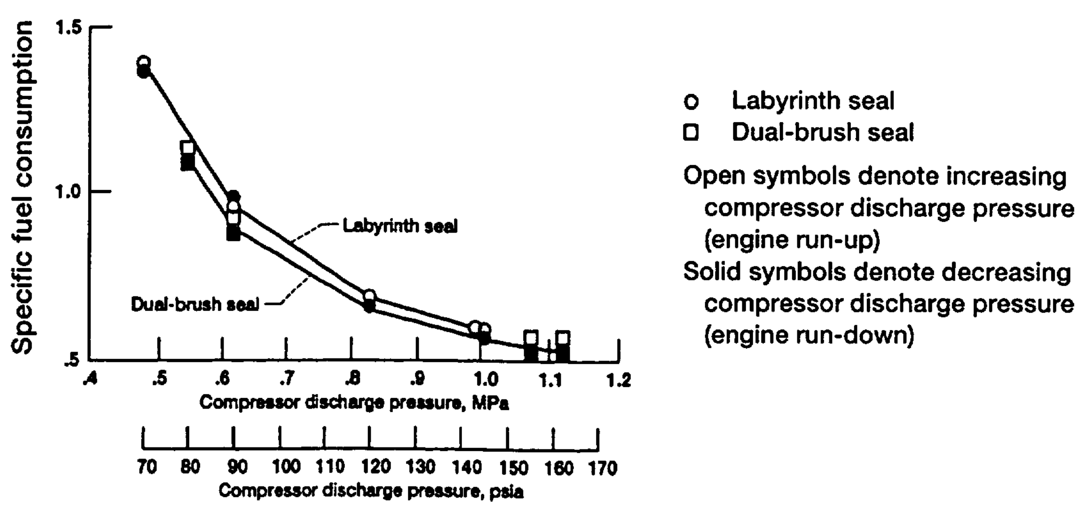

Figure 1.—Specific fuel consumption of experimental testbed engine as a function of compressor discharge pressure with labyrinth and dual-brush seals.

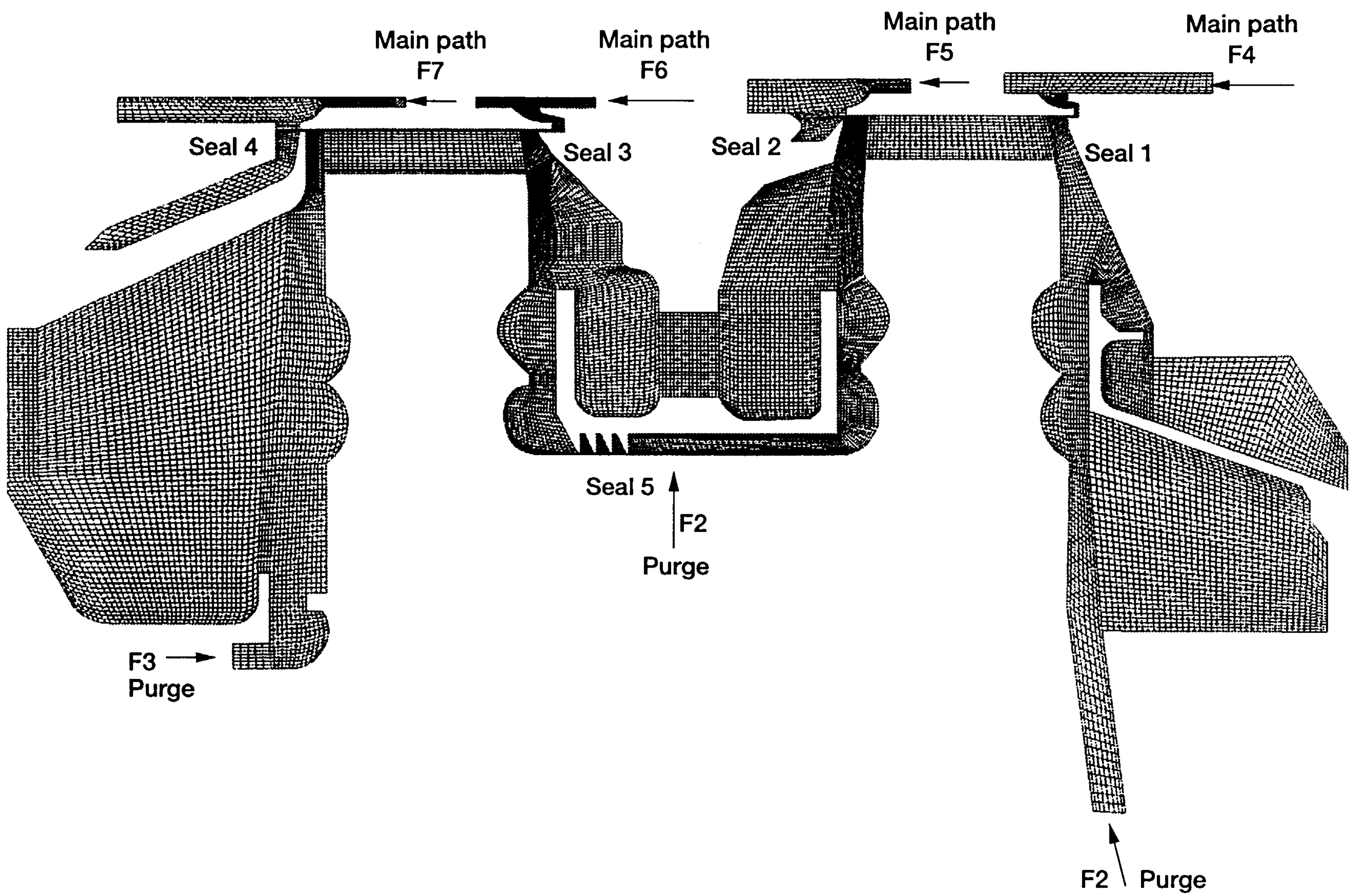

Figure 2.—SSME HPFTP grid, main gas path, and purge flows.

Contour levels	
3	−6.540E−02
6	−5.850E−02
9	−5.160E−02
12	−4.470E−02
15	−3.780E−02
18	−3.090E−02
21	−2.400E−02
24	−1.710E−02
27	−1.020E−02
30	−3.300E−03
33	3.600E−03
36	1.050E−02
39	1.740E−02

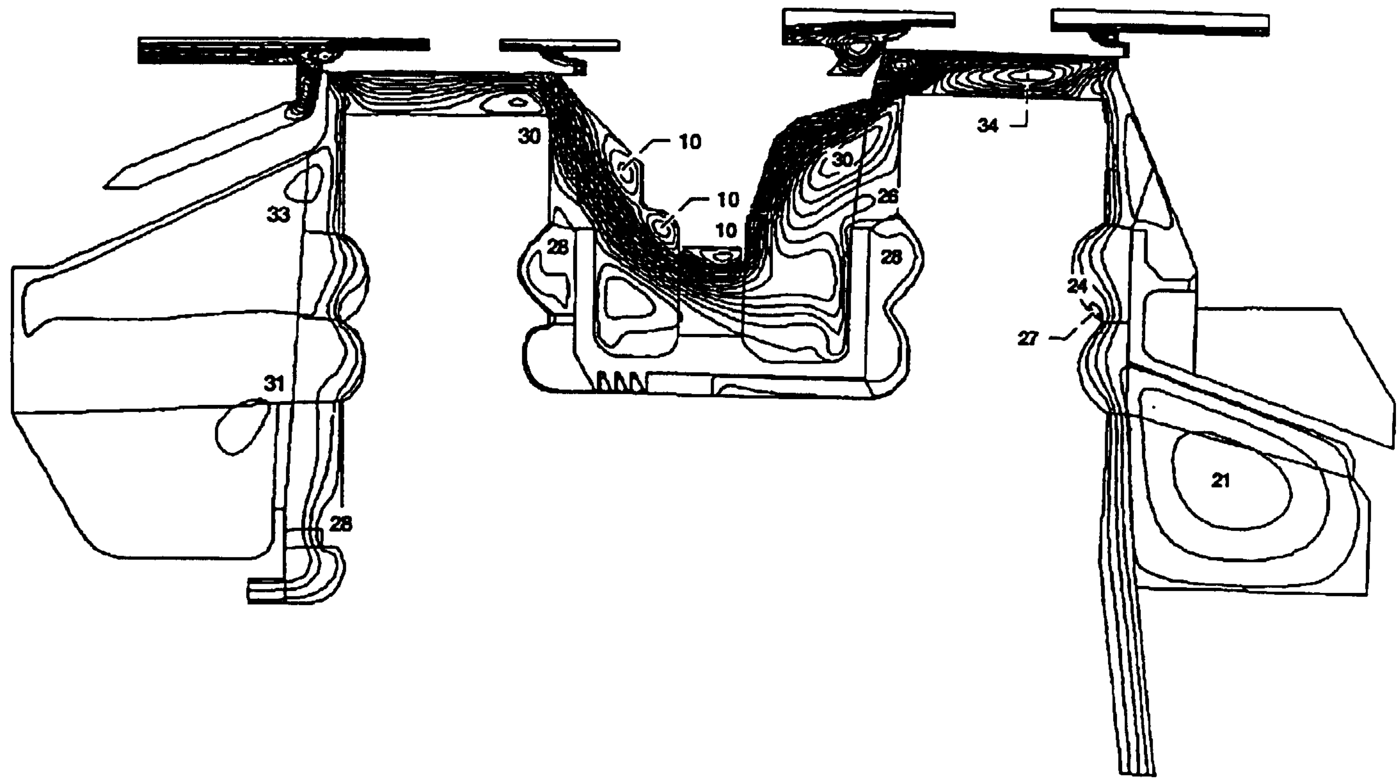

Figure 3.—Streamline patterns in multiply connected cavities of UTRC simulation of SSME HPFTP. For purge and main flows see figure 2.

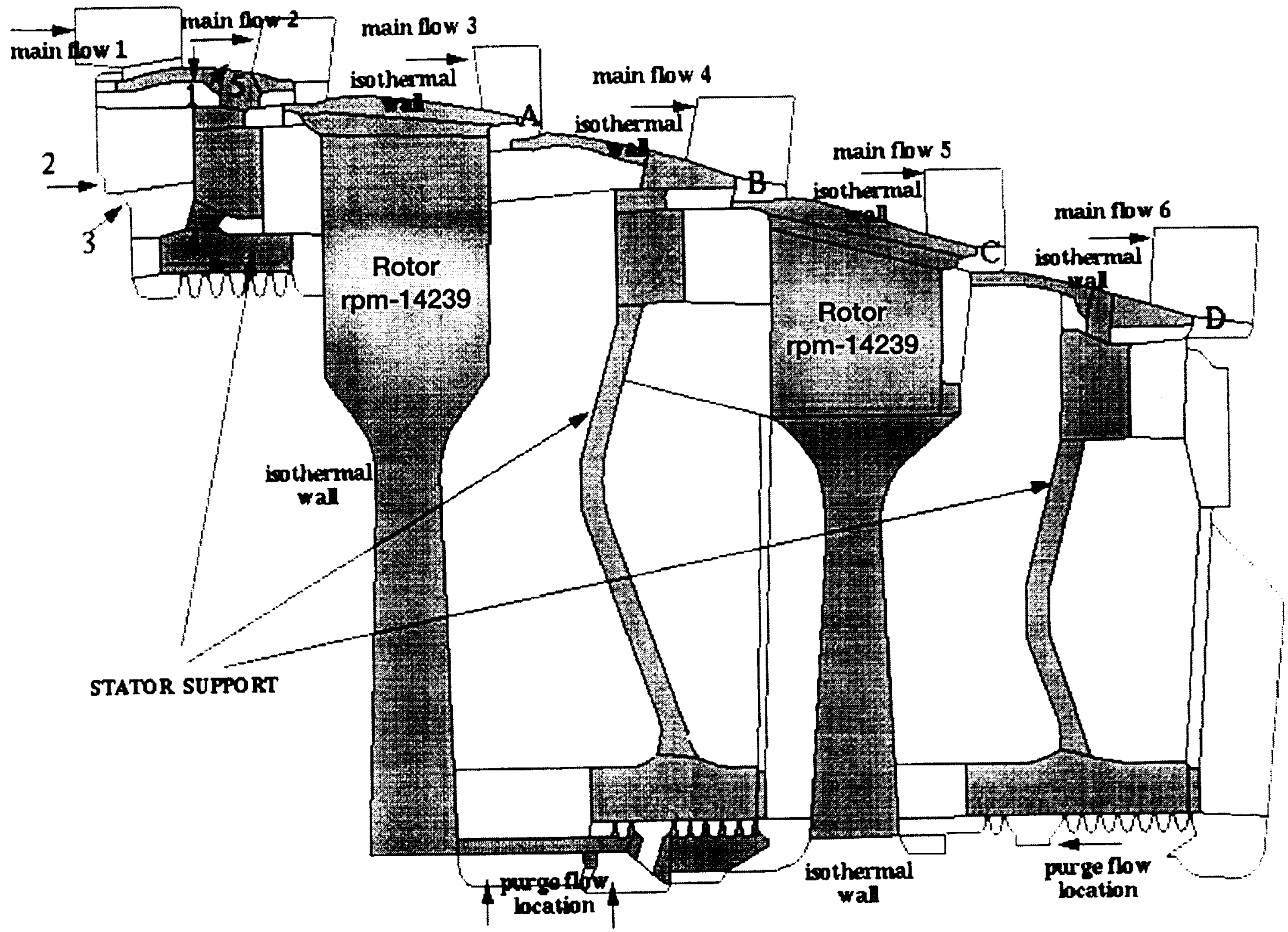

Figure 4.—Flow model for Allison T–56/501D turbine cavities. Shaded areas denote conjugate heat transfer. Static pressure is specified at six main flow exits.

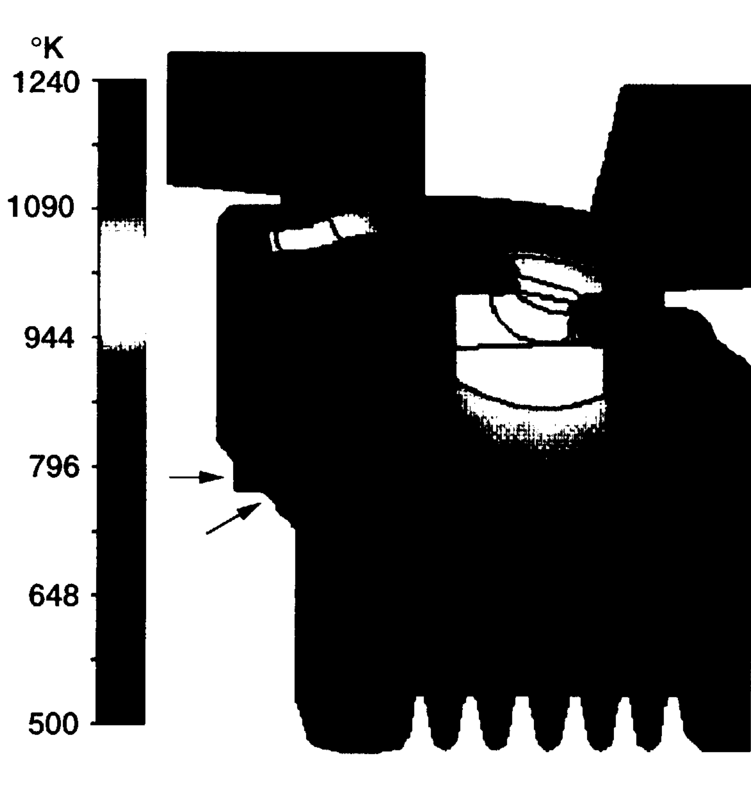

Figure 5.—Static temperature distribution inside stage 1–2 cavities.

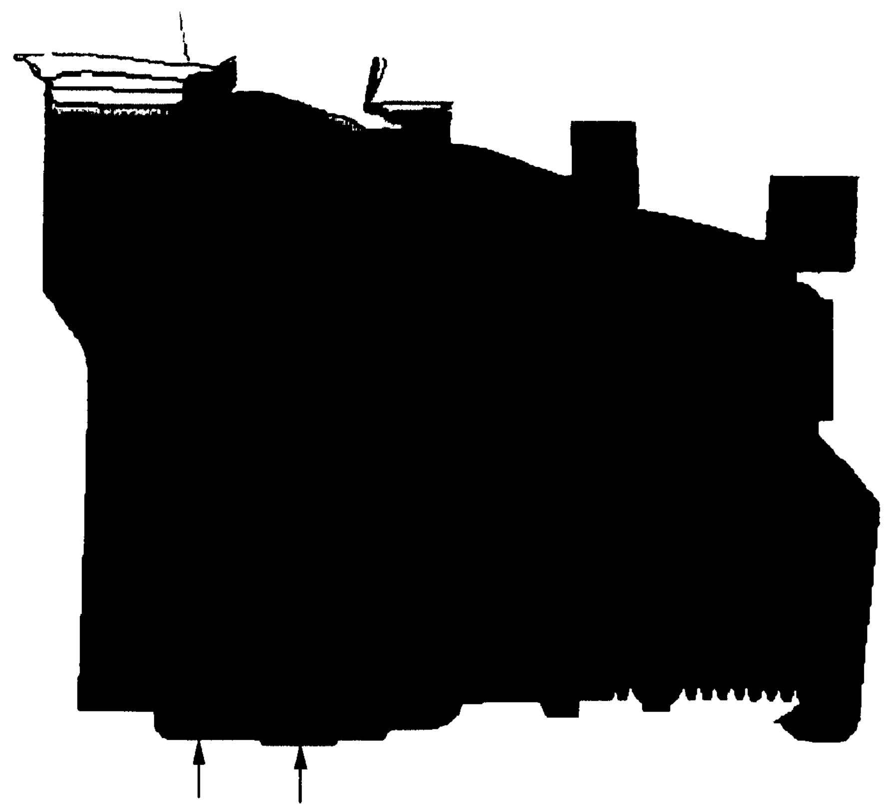

Figure 6.—Static temperature distribution inside stages 2–3 and 3–4 cavities.

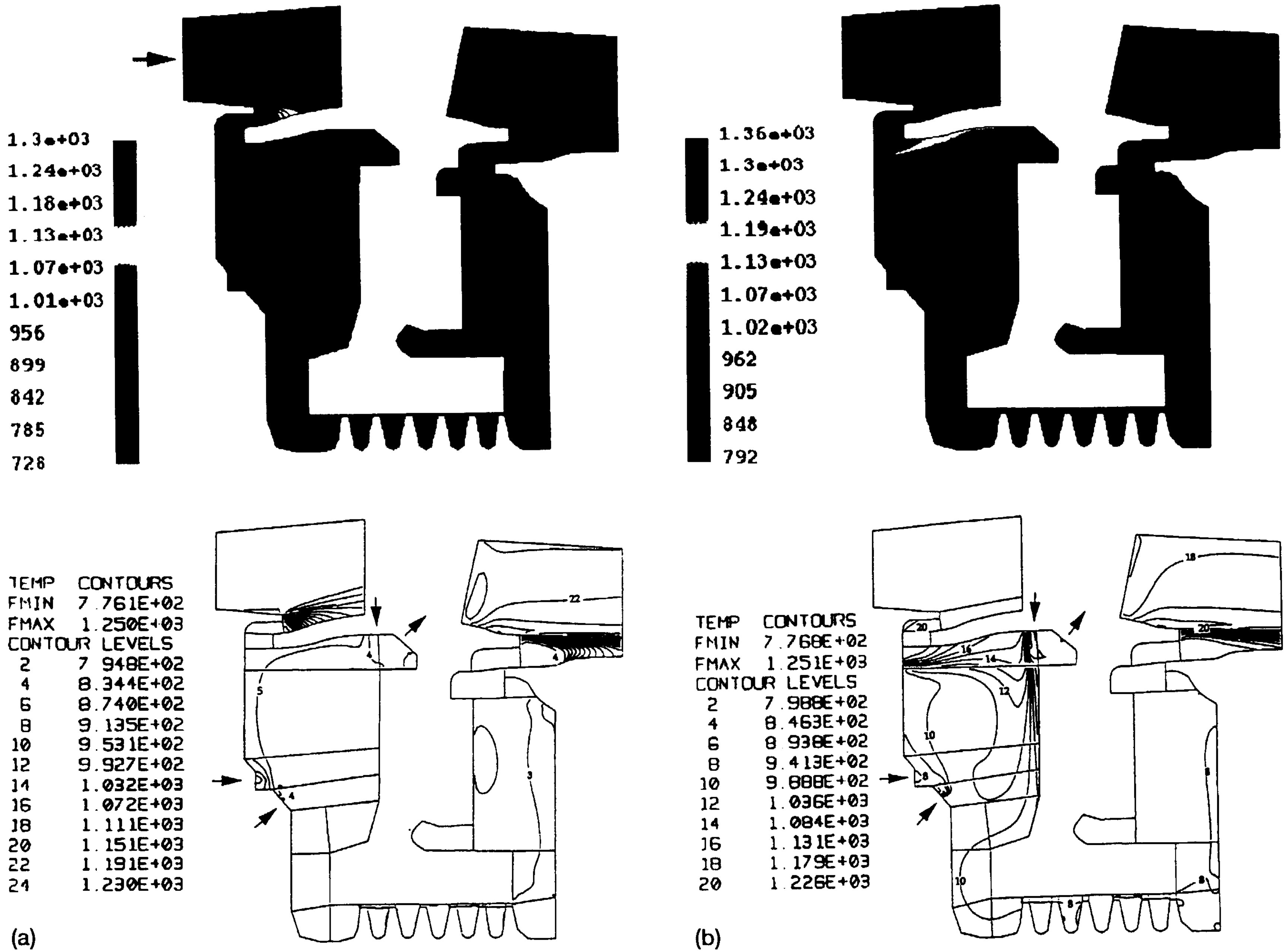

Figure 7.—Interstage turbine seal gas temperature contours at two clearances. (a) Labyrinth seal clearance, 0.012 in. (b) Labyrinth seal clearance, 0.024 in.

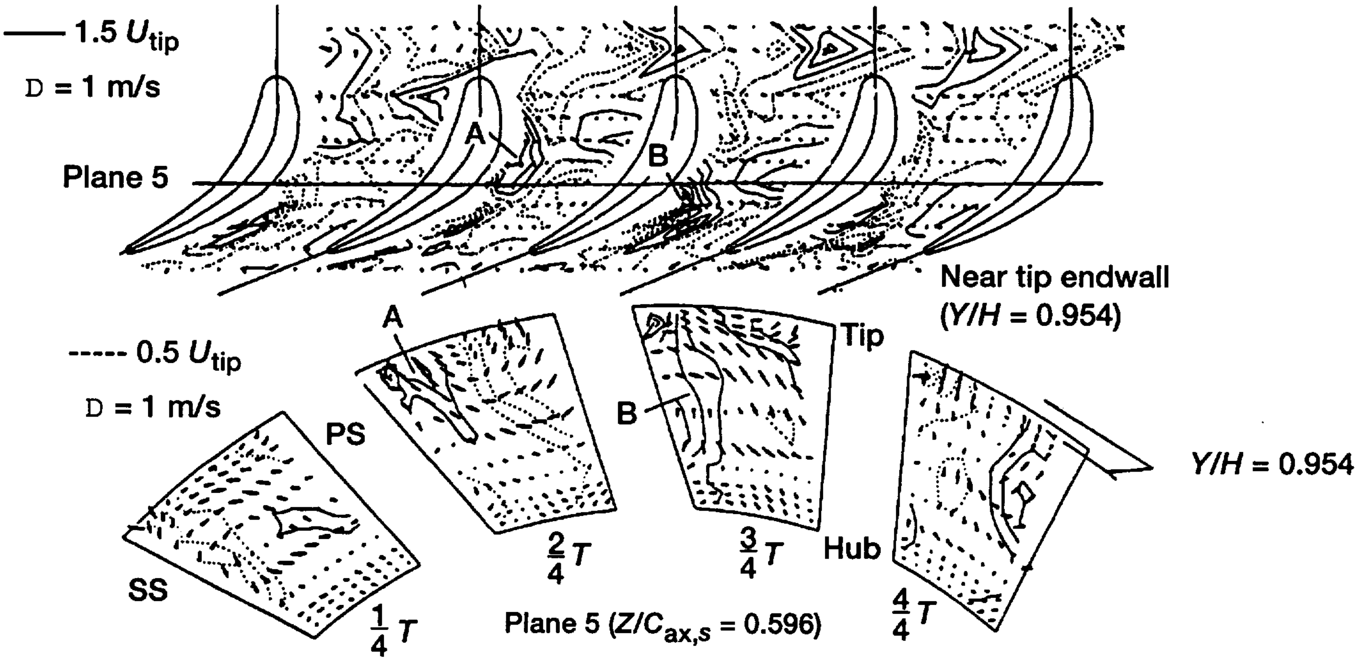

Figure 8.—Fluctuation of unsteady velocity within stator passages near tip endwall and plane 5. Yamamoto et al. 1993.

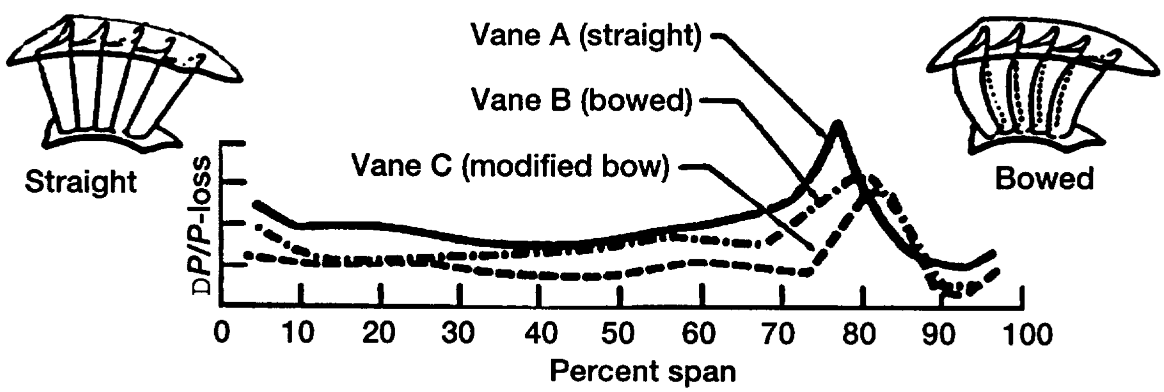

Figure 9.—Spanwise total pressure loss distribution for straight and bowed compressor blades. Huber et al. (1985).

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